DESIGN AND THERMAL-HYDRAULIC OPTIMIZATION OF A SHELL AND TUBE HEAT EXCHANGER USING BEES ALGORITHM

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The present study modifies the structural design of a shell-and-tube heat exchanger (STHE) by considering two key parameters such as the maximization of the overall heat transfer coefficient and minimization of the total pressure drop. Five geometric design variables which include the tube inside diameter, tube outside diameter, pitch size, baffle spacing, and the tube length are investigated for optimization. The governing equations for design and optimization of the STHE are evaluated; and the optimum design parameters are obtained by Bees Algorithm (BA). The selection of the important design parameters to achieve the proper design is evaluated by fixing each of these parameters, while the other the design parameters are selected as variable to optimize the effectiveness. Compared with the original STHE, the overall heat transfer coefficient is increased by 22.78 % with the minimum increase in the total pressure drop by 1.8%.

Key words*: shell-and-tube heat exchanger, overall heat transfer coefficient, pressure drop, Bees Algorithm*

1. Introduction

Heat exchangers are widely employed in engineering applications, including chemical industries, power generation, food industry, environmental engineering, energy recycling, air conditioning, and refrigeration systems. Today, with respect to advancements in technology, such as industrial processes, the need for high-efficient exchangers is felt more than ever. In recent studies, to optimize procedural and geometrical parameters and attain the maximum efficiency and minimum pressure drop in heat exchangers, the researchers have investigated geometrical, thermal, and hydraulic relations that are associated with each one of these exchangers and appropriately select the optimization algorithm as a new approach to the efficiency enhancement of heat exchangers [1-6]. Zarea *et al.* [7] have optimized a plate-fin heat exchanger by considering 7 parameters of optimization (the hot and cold inflow length, number of fin layers, fin frequency, fin height, fin length, and fin thickness) and by maximizing the efficacy of the exchanger and minimizing the entropy production through the ε-NTU (Number of Transfer Units) method and the Bees Algorithm. They have demonstrated that the utilization of the Bees Algorithm in the optimization of the plate-fin heat exchanger is efficient. Raja *et al.* [8] optimized an plate heat exchanger by considering eight design parameters, including the port diameter, horizontal and vertical distances between the ports, length of compact plates, plate thickness, chevron angle, enlargement factor, and the number of plates, and two objective functions, including the overall heat transfer coefficient and total pressure drop. Mirzaei *et al.* [9] have accomplished the optimization of the efficacy and the cost parameters in a shell-and-tube heat exchanger through a multi-objective optimization method with two objective functions of maximizing the efficacy and minimizing the cost. Thermal efficiency has increased by nearly 28 % by mixing the genetic algorithm with constructal theory. Zhicheng *et al.* [10] optimized a welded plate heat exchanger with straight gas channels and corrugated water channels using the grey correlation theory and computational fluid dynamics simulation and considering three design variables with H, A, and B dimensions. Their results show that the optimum dimension of this exchanger for attaining the maximum heat transfer coefficient of 70W/ (m^2°C) and the minimum pressure drop of 30 pa are as A=30mm, B/A=0.6, and H=24. Using the Non-dominated Sorting Genetic Algorithm (NSGA-II), Maghsoudi *et al.* [11] optimized four types of recuperative heat exchangers with rectangular, triangular, louver and offset fins in 200 kW microturbines, with recuperator efficiency, exergy efficiency, and total cost as the objective functions, the geometrical dimensions of the recuperator as design parameters, and the pressure drop and Reynolds number as optimization constraints. The optimization results show that the maximum thermal efficiency of the cycle and exergy and NPV (Net Percent Value) are occurred in the counter-flow recuperator with offset strip fin. Alimoradi [12] investigated the effect of effective and geometrical parameters on the exergy efficiency of a shell helical-coil heat exchanger. Their results displayed that, the efficiency of the coil which has the minimum diameter and maximum number of turns is higher when compared to the coils which have the same length. Zhang *et al.* [13] designed a novel heat exchanger, which preheats the air by the extra heat of the output gases, in high-temperature conditions to raise the heat recovery efficiency. In this research, an iterative algorithm was presented based on segmented LMTD (Logarithmic Mean Temperature Difference) method for design optimization. In their optimized exchanger, the efficiency of the heat transfer and NTU increased by 12.5% and 53%, respectively, compared to the common heat exchangers, while the pressure drop increased by 70% and 22%, respectively, in the air-side and gas-side. For the thermo-hydraulic optimization of a corrugated tube heat exchanger with V-cut and U-cut twisted tap inserts, Hassanpour *et al.* [14] employed the artificial neural networks together with genetic algorithm. In their research, the average difference between the prediction of the artificial neural network prediction and empirical results was small and around 2%. Their results showed that the corrugated tube with V-cut twisted tape has the maximum heat transfer rate. Many studies that address the heat exchanger design using optimization algorithms have achieved design parameters by disregarding the sensitivity of these parameters to objective functions. However, this study earmarks the sensitivity of the design parameters to heat transfer and pressure drop in determining appropriate selection parameters. The present study modifies the structural design of a shell-and-tube heat exchanger (STHE), illustrated in Fig. 1 with the stated characteristics in Table 1, by combining the thermo-hydraulic modeling and Bees optimization algorithm. The performance of this algorithm and the governing equations for the design and optimization of this heat exchanger are probed in the following sections.

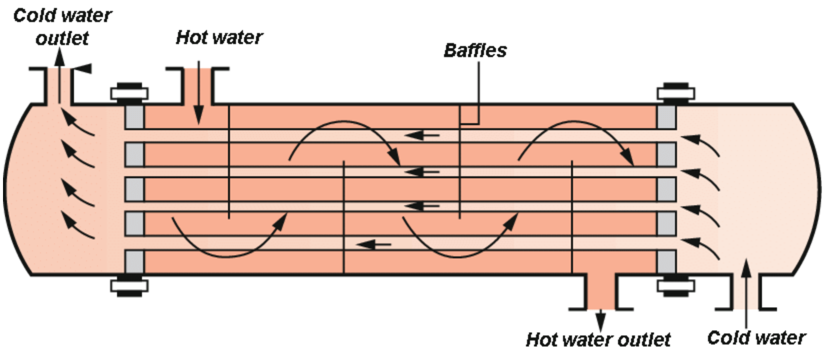


Figure 1. Simple schematic of a shell-and-tube heat exchanger

Table 1. The characteristics of the shell and tube heat exchanger [15]

Tube inside diameter (d_i)	16 mm
Tube outside diameter (d_o)	19 mm
Tube length (L)	4.54 m
Number of tubes (N_t)	374
Pitch size (P_t)	0.0254 m
Shell inside diameter (D_s)	0.58 m
Baffle spacing (B)	0.5 m

2. Bees Algorithm

The Bees Algorithm (BA) is a population-based search algorithm that has been developed by Pham *et al.* [16]. In this method, n number of bees are in search of a food/flower source (the solution of the problem) and each time that an artificial bee reaches the flower, the profit is evaluated. The bees possess this ability to improve the solution and find better ones by utilizing the others' information. This algorithm can be used to solve problems that have many solutions, some of which are better than other. So that, it starts with a random solution, and iteratively makes small changes to the solution, each time improving it a little. When the algorithm cannot see any improvement anymore, it terminates. In this study, the BA is selected to optimize the STHE due to the higher speed and accuracy in converging to the optimum value.

Table 2. Thermophysical properties of shell fluid and tube fluid

Property	Shell-side fluid	Tube-side fluid
c_p [J kg^{-1}K^{-1}]	4179	4182
ρ [kg m^{-3}]	995.9	998.2
k [Wm^{-1}K^{-1}]	0.612	0.598
μ[kg m^{-1} s^{-1}]	8.15×10^{-4}	10.02×10^{-4}

3. Modeling Formulation

This section describes thermal-hydraulic modelling of the STHE, objective function formulation, design variables, and constraints involved in STHE design optimization.

3.1. Thermal and hydraulic formulation

The current study estimates the heat transfer and pressure drop coefficients by considering the thermo-physical properties of the cold water (tube fluid) and hot water (shell fluid), in the thermo-hydraulic modeling of the shell-tube heat exchangers for the shell and tube, presented in Table 2. The hot water at 305 K is entering into STHE with the mass flow rate of 50 kg/s. The cold water having the mass flow rate of 150 kg/s is supplied to STHE at a temperature of 293 K. The STHE is assumed to running under a steady state, with negligible heat loss and uniform velocities. Further, heat transfer coefficients are assumed to be uniform and constant. The tubes are single-pass in this heat exchanger.

3.1.1 Shell side calculation

(a) The heat transfer coefficient of the shell fluid is calculated by the below relation for the Reynolds number range of $2\times10^3 < \mathrm{Re}_h < 10^6$ [17]:

$$h_h = \frac{0.36 k_h}{D_e} Re_h^{0.55} Pr_h^{\frac{1}{3}} \tag{1}$$

In the Eq. (1), the shell equivalent diameter (D_e) is defined as the below equation:

$$D_e = \frac{4\left(P_t^2 - \frac{\pi d_o^2}{4}\right)}{\pi d_o} \tag{2}$$

where d_o and P_t are the tube outside diameter and pitch size, respectively.

In the Eq. (1), the Reynolds and Prandtl numbers of the shell fluid are computed as the following:

$$Re_h = \left(\frac{\dot{m}_h}{A_s}\right)\frac{D_e}{\mu_h} \tag{3}$$

$$Pr_h = \frac{c_{p,h}\mu_h}{k_h} \tag{4}$$

In Eq. (3), $\dot{m}_h$ is the mass flow rate of the shell fluid, and A_s is the crossflow area and calculated by the below relation:

$$A_s = (D_s - N_{TC} d_o)B \tag{5}$$

$$N_{TC} = \frac{D_s}{P_t} \tag{6}$$

In Eq. (5), D_s and B are the shell inside diameter and the baffle spacing, respectively. D_s is calculated as below [15]:

$$D_s = \left(\frac{N_t (CL)(PR)^2 d_o^2}{0.785(CTP)}\right)^{0.5} \tag{7}$$

PR is the tube pitch ratio (P_t/d_o). Also, *CL* and *CTP* are the tube layout constant and the tube count calculation constant. *CTP* equals 0.93 in the STHE with single-pass tubes, and *CL* equals 1 in the square arrangement of the tubes.

In Eq. (7), having the mass flow rate of the tube fluid and considering the maximum velocity of u_m=2 m^2/s to preclude tube erosion, we can estimate the number of tubes (N_t) by the below relation [15]:

$$N_t = \frac{4\dot{m}_c}{\rho_c \pi d_i^2 u_m} \tag{8}$$

(b) The pressure drop in the shell (hot flow) is calculated by the below relation for $400 < \mathrm{Re} \leq 10^6$ [15]:

$$\Delta p_S = f_h \left(\frac{\dot{m}_h^2 D_s (N_b + 1)}{2\rho_h A_s^2 D_e} \right) \tag{9}$$

In Eq. (9), the friction coefficient and the number of baffles are equal to:

$$f_h = \exp(0.576 - 0.19 \ln \mathrm{Re}_h) \tag{10}$$

$$N_b = \frac{L}{B} - 1 \tag{11}$$

3.1.2 Tube side calculation

(a) The heat transfer coefficient of the tube fluid is calculated by defining the Nusselt Petokhov number (for Re_c>10000) [18] as follows:

$$h_c = \frac{Nu_c k_c}{d_i} \tag{12}$$

$$Nu_c = \frac{(f_c / 2)\,\mathrm{Re}_c\,\mathrm{Pr}_c}{1.07 + 12.7(f_c / 2)^{1/2}(\mathrm{Pr}_c^{1/2} - 1)} \tag{13}$$

In Eq. (13), the friction coefficient (for Re_c >10000), Reynolds number, and Prandtl number for the tube fluid, respectively, equal [15]:

$$f_c = (1.58 \ln \mathrm{Re}_c - 3.28)^{-2} \tag{14}$$

$$\mathrm{Re}_c = \frac{\rho_c u_m d_i}{\mu_c} \tag{15}$$

$$Pr_c = \frac{c_{p,c} \mu_c}{k_c} \tag{16}$$

(b) The pressure drop in the tube path (Δp_t) and the pressure drop due to the change of direction in the passes (Δp_r) are computed by the following equations [15]:

$$\Delta p_t = \frac{2 f_c N_p L \rho_c u_m^2}{d_i} \tag{17}$$

$$\Delta p_r = 2 N_p \rho_c u_m^2 \tag{18}$$

Thus, the total pressure drop in tubes equals:

$$\Delta p_T = \Delta p_t + \Delta p_r \tag{19}$$

The tubes in this heat exchanger are single-pass (N_p = 1).

3.1.3 Overall heat transfer coefficient

By calculating the heat transfer coefficients of the tube and shell fluids, we can estimate the overall heat transfer coefficient based on the below equation [15]:

$$U = \left(\frac{1}{h_h} + \frac{1}{h_c} \times \frac{d_o}{d_i} + \frac{r_o \ln(r_o / r_i)}{k} \right)^{-1} \tag{20}$$

In Eq. (20), the thermal conductivity of the tube material (Cr-alloy) is equal to k= 42.3 W/ (m K).

3.2. Objective function and design parameters

In the BA, the ratio of the overall heat transfer coefficient to the total pressure drop is considered as an objective function. The first is the maximization of the overall heat transfer coefficient and second target is the minimization of the pressure drop to decrease the operating cost. The population (number of bees) is 30, and the iteration number is 100 in this optimization algorithm. In order to cover the majority design range of the STHE, the design variables including the inside and outside diameters of the tube, pitch size, baffle spacing, and tube length are considered in their allowable range with regard to the dimensions restrictions of the STHE designed by Kakac *et al.* [15].

Table 3. Design parameters and corresponding ranges

Decision Variables	Range
d_i *(m)*	0.1-0.3
d_o *(m)*	0.0003-0.001
P_t *(m)*	1.15-1.25
B (m)	0.3-0.6
L (m)	0.3-0.7

4. Results and Discussion

Using the Bees Algorithm, this study optimized five design parameters in a shell-tube heat exchanger to achieve the maximum overall heat transfer coefficient with the minimum possible pressure drop and presented the optimized values in Table 4. As observed in the table, by selecting the (1) and (2) solution sets of the optimum design parameters from the solution sets of the Bees Algorithm output, as the most suitable design parameters of the STHE, the overall heat transfer coefficient increases by 22.78% and 1%, and the total pressure drop increases by 1.8% and decreases by 25%, compared to the results of Kakac *et al.* [15]. Therefore, the designer of the heat exchanger can choose the optimized design parameters based on the requirements of the corresponding unit for purposes of either increasing the transferred heat exchange or the pressure drop decrease. An increase in the heat transfer of heat exchangers is often accompanied by a noticeable increase in the pressure drop. However, by selecting the design parameters of the solution set (1) from Table 4, we witness a proper increase in the heat transfer with the minimum pressure drop of about 1.8%, which is very desirable. Furthermore, as presented in Table 5, the number of tubes, the number of baffles, and shell diameter are specified by the determination of the optimum design parameters of the solution sets (1) and (2) in the optimization process of the heat exchanger. The results reveal that the numbers of tubes and baffles and shell diameter decrease during the optimization of the STHE in the present study as compared with the heat exchanger designed by Kakac *et al.* [15]. Finally, in order to choose the appropriate design parameters, this paper numerically investigates the effects of five design parameters on the overall heat transfer coefficient and total pressure drop by fixing one of these parameters and considering that the other four parameters are variable.

Table 4. The comparison between the design results and the corresponding results from reference [15]

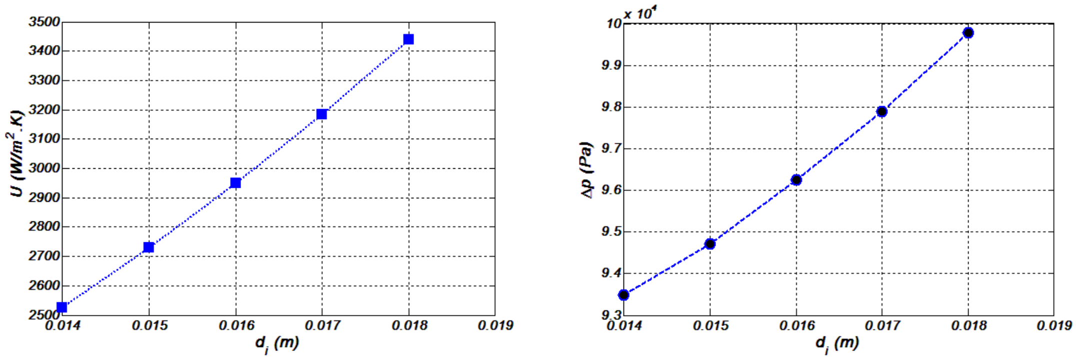
	Reference [15]	BA (1)	BA (2)
d_i *(m)*	0.016	0.017392	0.017524
d_o *(m)*	0.019	0.018782	0.018675
P_t *(m)*	0.0254	0.02413	0.025784
B (m)	0.5	0.52116	0.55683
L (m)	8	6.6484	7.472
U ($Wm^{-2}K^{-1}$)	2458.1	3018.1	2480.6
ΔP (Pa)	55727	56733	41610

Table 5. The comparison between the value of N_t, N_b and D_s obtained by BA and the corresponding results from reference [15]

	Reference [15]	BA (1)	BA (2)
N_t	374	317	312
N_b	15	12	13
D_s (m)	0.5749	0.5028	0.533

4.1. The impact of the tube inside diameter on the optimization process

The variations in the overall heat transfer coefficient and total pressure drop in the STHE, when the design parameter of the tube inside diameter is constant and the other four parameters are variable, are pictured in Figure 2. The results show that the overall heat transfer coefficient properly increases by 36.2%, and the total pressure drop slightly increases by 6.72% as the tube inside diameter rises from 0.014 m to 0.018 m. The optimal values of the other four design variables in this optimization process have been similarly selected and presented in Table 6.



(a) **(b)**

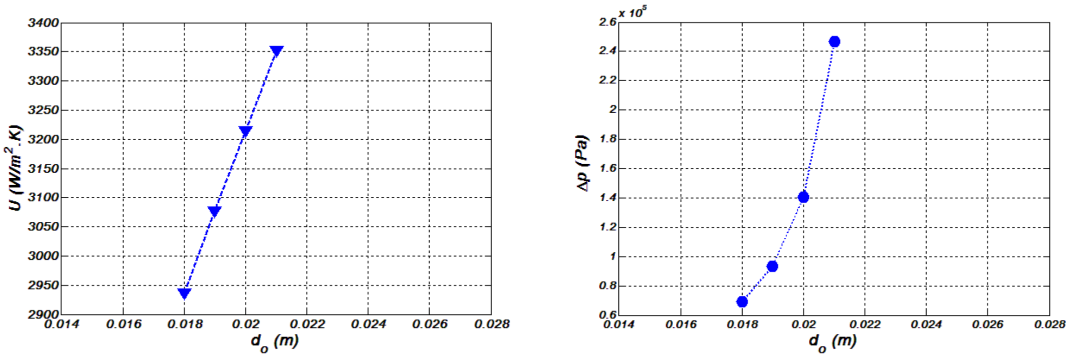
Figure 2. The impact of the tube inside diameter on (a) overall heat transfer coefficient (b) total pressure drop

Table 6. The optimal variables in the optimization process at different tube inside diameter

L (m)	*B* (m)	P_t *(m)*	d_o *(m)*
7.6716	0.41605	0.02408	0.018782

4.2. The impact of the tube outside diameter on the optimization process

Figure 3 displays that the overall heat transfer coefficient increases by 14.14% during the optimization of the STHE when the tube outside diameter is constant in the range of 0.018 m to 0.021 m, while the total pressure drop experiences a remarkable 3.56-time increase. Thus, the results reveal the inappropriate design of the STHE when the design parameter of the tube outside diameter is constant. The optimal values of the four design parameters have been similarly selected in the optimization process and presented in Table 7.



(a) (b)

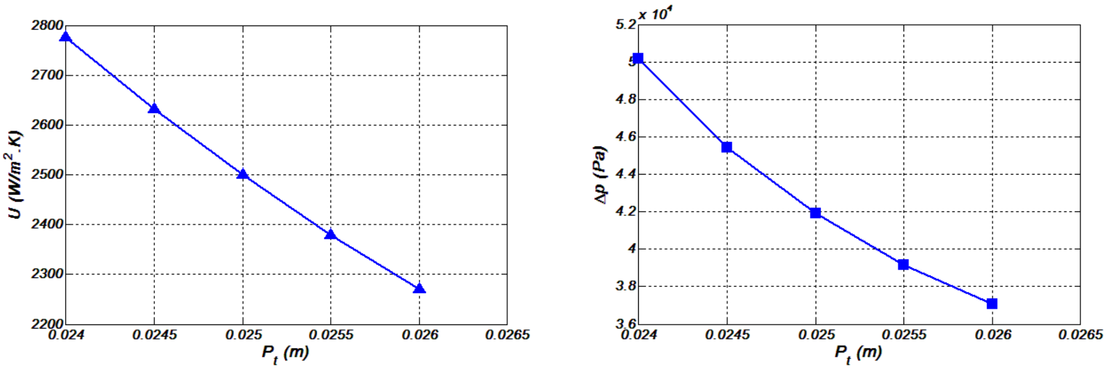
Figure 3. The impact of the tube outside diameter on (a) overall heat transfer coefficient (b) total pressure drop

Table 7. The optimal variables in the optimization process at different tube outside diameter

L (m)	B (m)	P_t (m)	d_i (m)
7.9852	0.42525	0.024397	0.016947

4.3. The impact of the pitch size on the optimization process

Figure 4 depicts that the overall heat transfer coefficient and total pressure drop decrease by 18.22% and 26.11%, respectively, when the pitch size is constant in the range of 0.024 to 0.026, and the other four design parameters are variable. Hence, although the total pressure drop decreases properly when the pitch size is constant, the constancy of this design parameter is not recommended in the STHE optimization due to a robust decrease in the overall heat transfer coefficient. The optimal values of the four parameters have been similarly selected in the optimization process and presented in Table 8.



(a) (b)

Figure 4. The impact of the pitch size on (a) overall heat transfer coefficient (b) total pressure drop

Table 8. The optimal variables in the optimization process at different pitch size

L (m)	B (m)	d_o (m)	d_i (m)
6.5794	0.5391	0.01832	0.016561

4.4. The impact of the baffle spacing on the optimization process

Figure 5 illustrates that the total pressure drop considerably decreases by 36.77% when the design parameter of baffle spacing is constant between 0.4 m and 0.6 m; however, the overall heat transfer coefficient decreases by 14.34%, too. Thus, the results show that we can attain a considerable decrease in the total pressure drop in the STHE by considering the baffle spacing as the design constant and the other four variables as the design variables. The optimal values of the four design parameters have been similarly selected and presented in Table 9. Comparing Tables 8 and 9, we witness that similar L, d_o, and d_i design variables have been selected in the optimization process when the pitch size and baffle spacing design parameters are constant.

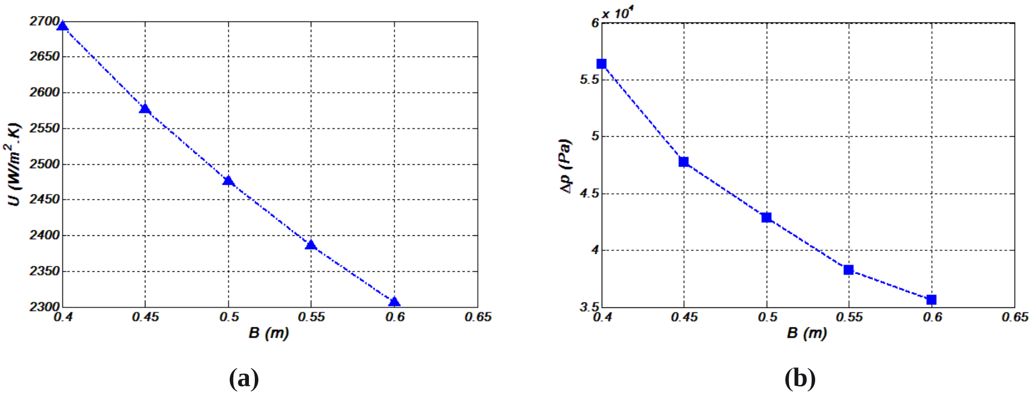


Figure 5. The impact of the baffe spacing on (a) overall heat transfer coefficient (b) total pressure drop

Table 9. The optimal variables in the optimization process at different baffle spacing

L (m)	P_t (m)	d_o (m)	d_i (m)
6.5794	0.025391	0.01832	0.016561

4.5. The impact of the tube length on the optimization process

The value of the overall heat transfer coefficient reached a fixed value of 2882 W/ (m^2 K) when the design parameter of the tube length was constant in the 6-8m range, and the other four design parameters were variable; however, the total pressure drop increased by 28.43%, as illustrated in Figure 6. The optimal values of the four design variables have been similarly selected in this optimization process and represented in Table 10. No suitable design is obtained during the optimization process of the STHE so the tube length is considered as a design variable in our calculations.

Table 10 The optimal variables in the optimization process at different tube length

B (m)	P_t (m)	d_o (m)	d_i (m)
0.53276	0.024454	0.019281	0.016981

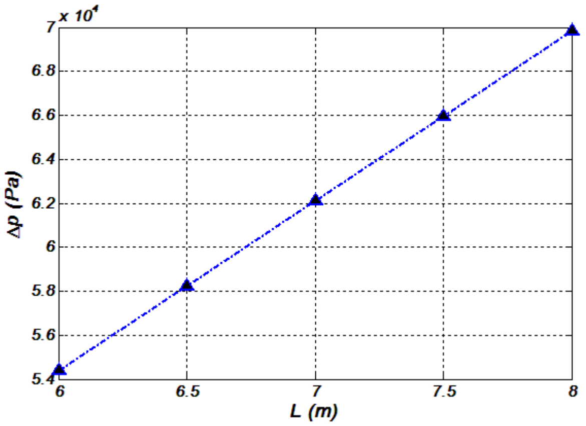


Figure 6. The impact of the tube length on the total pressure drop

5. Conclusion

An increase in the heat transfer of heat exchangers is usually accompanied by a noticeable increase in pressure drop. Using the Bees Algorithm, the present study modified five design parameters, including the tube inside diameter, tube outside diameter, pitch size, baffle spacing, and the tube length, in a shell-tube heat exchanger to achieve the maximum overall heat transfer coefficient with the least possible pressure drop. Likewise, to choose the appropriate parameters, it evaluated and analyzed the effects of the five design parameters on the shell-tube heat exchanger efficiency when one parameter was constant, and the other four parameters were variable. Finally, two optimal solution sets were selected from the output solution sets of the bee algorithm for the design parameters based on two objective functions, including the overall heat transfer coefficient and total pressure drop in the optimization process of the STHE. The overall heat transfer coefficient can increase by 22.78% with the minimum increase in the total pressure drop by 1.8%, and the pressure drop can decrease by 25% with the minimum increase in the overall heat transfer coefficient by 1%, when every one of these solution sets is selected.

Nomenclature

A_s	crossflow area, m^2
B	baffle spacing, m
c_p	specific heat of fluid, J kg^{-1} K^{-1}
D_e	equivalent diameter, m
D_s	shell inside diameter, m
d_o	tube outside diameter, m
d_i	tube inside diameter, m
f	friction factor
h	heat transfer coefficient, W m^{-2} K^{-1}
k	thermal conductivity, W m^{-1} K^{-1}
L	tube length, m
$\dot{m}$	mass flow rate, kg s^{-1}
N_b	number of baffles
N_t	number of tubes
Nu	Nusselt number
P_t	pitch size, m
Pr	Prandtl number
Re	Reynolds number
U	overall heat transfer coefficient, W m^{-2} K^{-1}

u_m average velocity inside tubes, m s^{-1}
Δp pressure drop, Pa

Greek Symbols

ρ fluid density, kg m^{-3}
μ viscosity, kg m^{-1} s^{-1}

Subscripts

c cold fluid
h hot fluid

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